

SCREW/THREADED FASTENERS



Formed by cutting a continuous helical groove on a cylindrical surface.

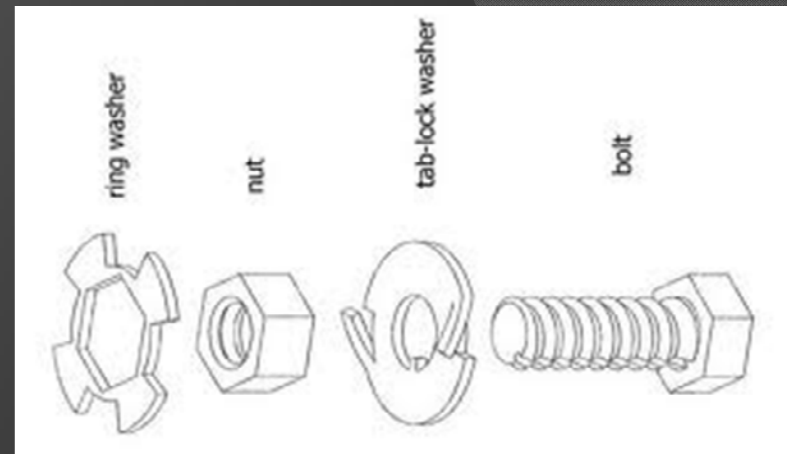
Widely used where the machine parts are required to readily connected and disconnected without damage to the machine or the fasteners.

Screwed fasteners:

1. Bolt
2. Nut
3. Washer/locking mechanism

Advantages:

1. Highly reliable
2. Convenient to assemble & disassemble.
3. Wide range of screwed joints may be adopted to various operating conditions.
4. Cheap to produce due to standardization and highly efficient manufacturing processes.



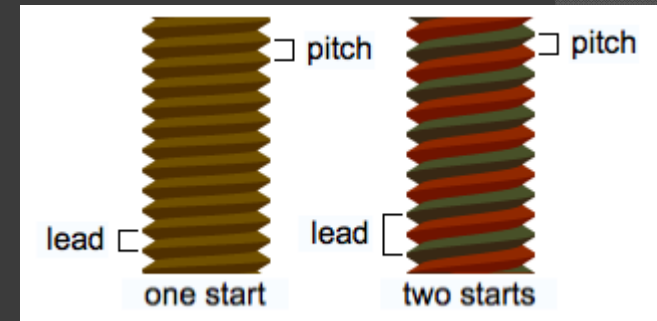
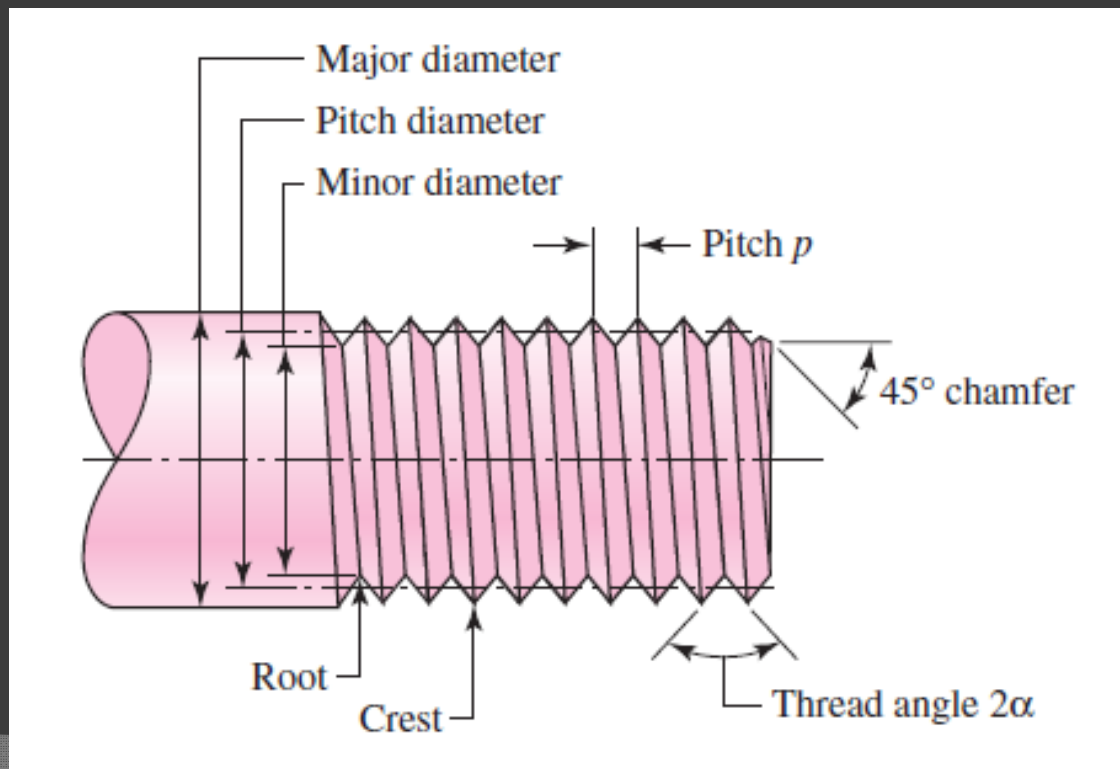
Disadvantages:

1. Stress concentrations in the threaded portions.

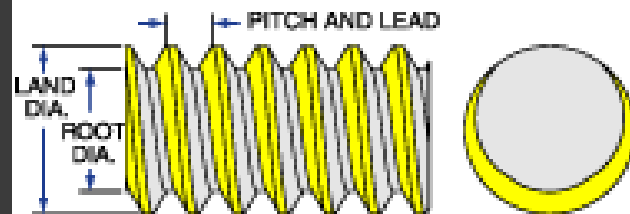
Note: Strength of screwed joints not to be compared with that of riveted or welded joints.

Material:

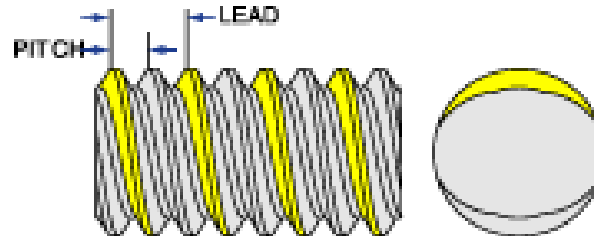
Screws and bolts are made from a wide range of materials, with [steel](#) being perhaps the most common, in many varieties. Where great resistance to weather or corrosion is required, stainless steel, [titanium](#), [brass](#) (steel screws can discolor oak and other woods), [bronze](#), monel or silicon bronze may be used, or a coating such as brass, [zinc](#) or [chromium](#) applied. Electrolytic action from dissimilar metals can be prevented with [aluminium](#) screws for double-glazing tracks, for example. Some types of plastic, such as [nylon](#) or [polytetrafluoroethylene](#) (PTFE), can be threaded and used for fastening requiring moderate strength and great resistance to corrosion or for the purpose of electrical [insulation](#)



**SINGLE START
(LEAD = PITCH)**



**DOUBLE START
(LEAD = 2 x PITCH)**



**FOUR START
(LEAD = 4 x PITCH)**

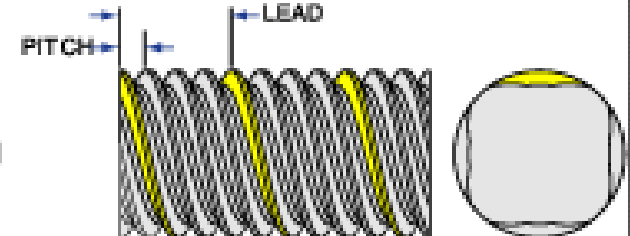


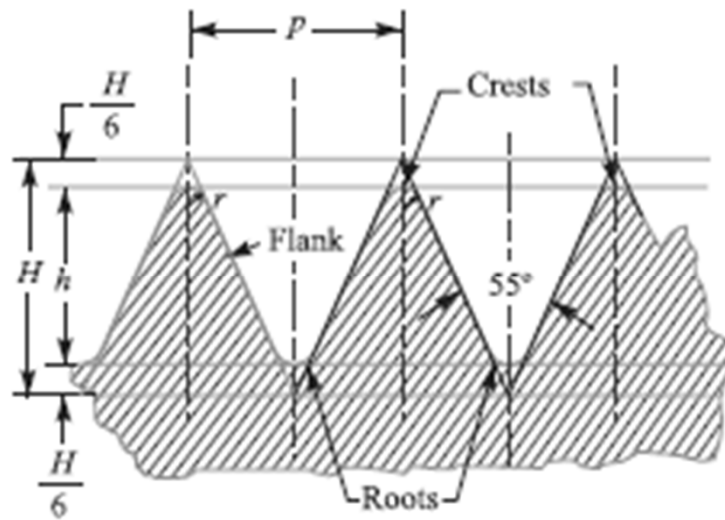
Table 7.1 Basic dimensions for ISO metric screw threads (coarse series)

Designation	Nominal or major dia d/D (mm)	Pitch (p) (mm)	Pitch diameter d_p/D_p (mm)	Minor diameter		Tensile stress area (mm ²)
				d_c	D_c	
M 4	4	0.70	3.545	3.141	3.242	8.78
M 5	5	0.80	4.480	4.019	4.134	14.20
M 6	6	1.00	5.350	4.773	4.917	20.10
M 8	8	1.25	7.188	6.466	6.647	36.60
M 10	10	1.50	9.026	8.160	8.376	58.00
M 12	12	1.75	10.863	9.853	10.106	84.30
M 16	16	2.00	14.701	13.546	13.835	157
M 20	20	2.50	18.376	16.933	17.294	245
M 24	24	3.00	22.051	20.319	20.752	353
M 30	30	3.50	27.727	25.706	26.211	561
M 36	36	4.00	33.402	31.093	31.670	817
M 42	42	4.50	39.077	36.479	37.129	1120
M 48	48	5.00	44.752	41.866	42.587	1470
M 56	56	5.50	52.428	49.252	50.046	2030
M 64	64	6.00	60.103	56.639	57.505	2680
M 72	72	6.00	68.103	64.639	65.505	3460
M 80	80	6.00	76.103	72.639	73.505	4340
M 90	90	6.00	86.103	82.639	83.505	5590
M 100	100	6.00	96.103	92.639	93.505	7000

Table 7.2 Basic dimensions for ISO metric screw threads (fine series)

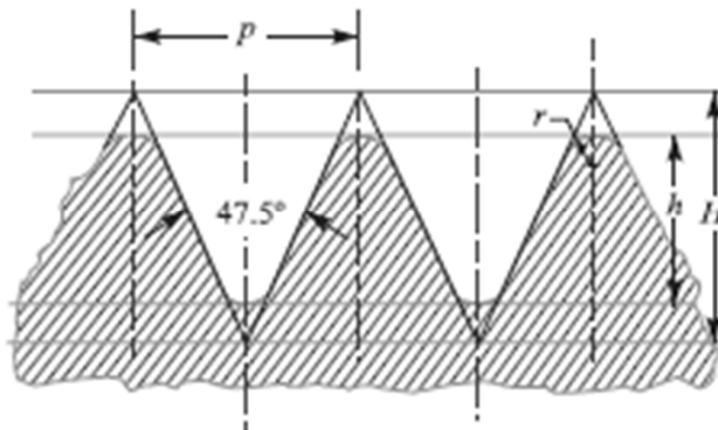
Designation	Nominal or major dia d/D (mm)	Pitch (p) (mm)	Pitch diameter d_p/D_p (mm)	Minor diameter		Tensile stress area (mm ²)
				d_c	D_c	
M 6 × 1	6	1.00	5.350	4.773	4.917	20.1
M 6 × 0.75	6	0.75	5.513	5.080	5.188	22.0
M 8 × 1.25	8	1.25	7.188	6.466	6.647	36.6
M 8 × 1	8	1.00	7.350	6.773	6.917	39.2
M 10 × 1.25	10	1.25	9.188	8.466	8.647	61.2
M 10 × 1	10	1.00	9.350	8.773	8.917	64.5
M 12 × 1.5	12	1.50	11.026	10.160	10.376	88.1
M 12 × 1.25	12	1.25	11.188	10.466	10.647	92.1
M 16 × 1.5	16	1.50	15.026	14.160	14.376	167
M 16 × 1	16	1.00	15.350	14.773	14.917	178
M 20 × 2	20	2.00	18.701	17.546	17.835	258
M 20 × 1.5	20	1.50	19.026	18.160	18.376	272
M 24 × 2	24	2.00	22.701	21.546	21.835	384
M 24 × 1.5	24	1.50	23.026	22.160	22.376	401
M 30 × 3	30	3.00	28.051	26.319	26.752	581
M 30 × 2	30	2.00	28.701	27.546	27.835	621
M 36 × 3	36	3.00	34.051	32.319	32.752	865
M 36 × 2	36	2.00	34.701	33.546	33.835	915
M 42 × 4	42	4.00	39.402	37.093	37.670	1150
M 42 × 3	42	3.00	40.051	38.319	38.752	1210
M 48 × 4	48	4.00	45.402	43.093	43.670	1540
M 48 × 3	48	3.00	46.051	44.319	44.752	1600

Forms of Screw Threads:



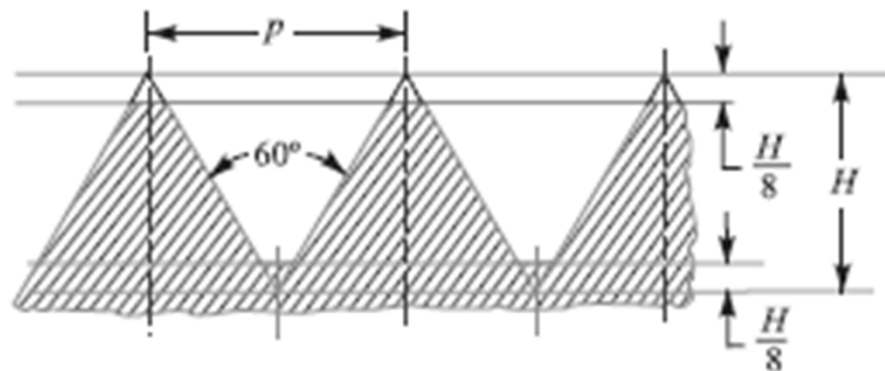
$$H = 0.96 p ; h = 0.64 p ; r = 0.1373 p$$

British standard whitworth (B.S.W) thread.



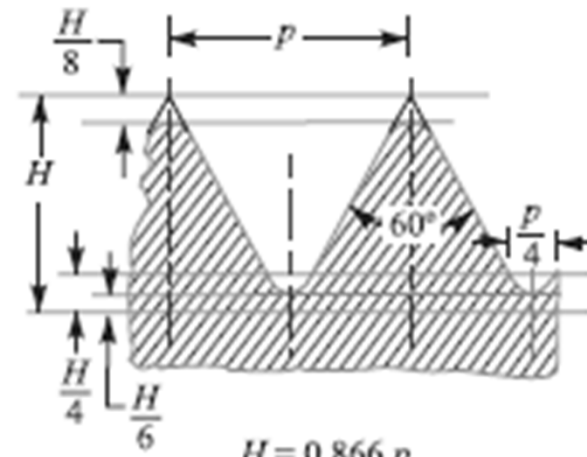
$$H = 1.13634 p ; h = 0.6 p ; r = 0.18083 p$$

British association (B.A.) thread.



$$H = 0.866 p$$

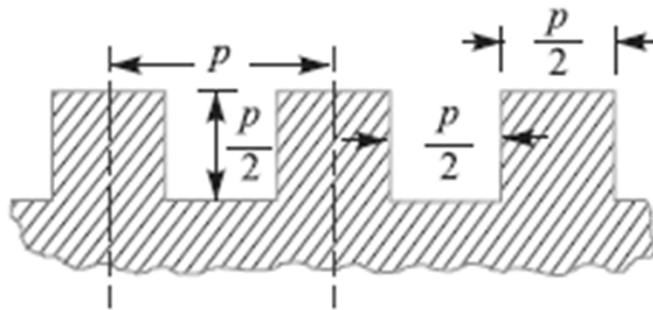
American national standard thread.



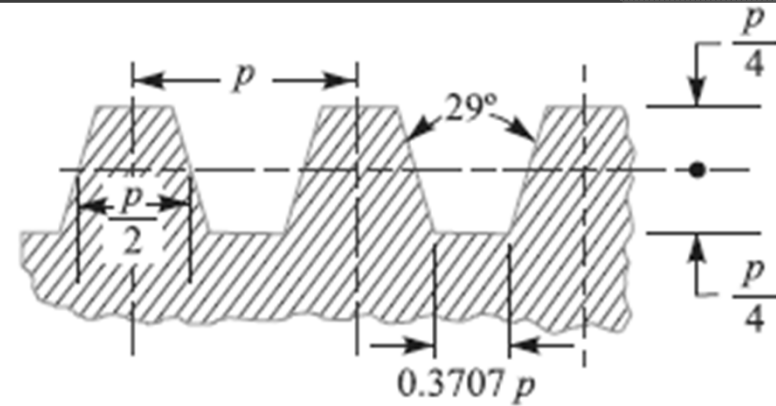
$$H = 0.866 p$$

Unified standard thread.

transmission of power in either direction.



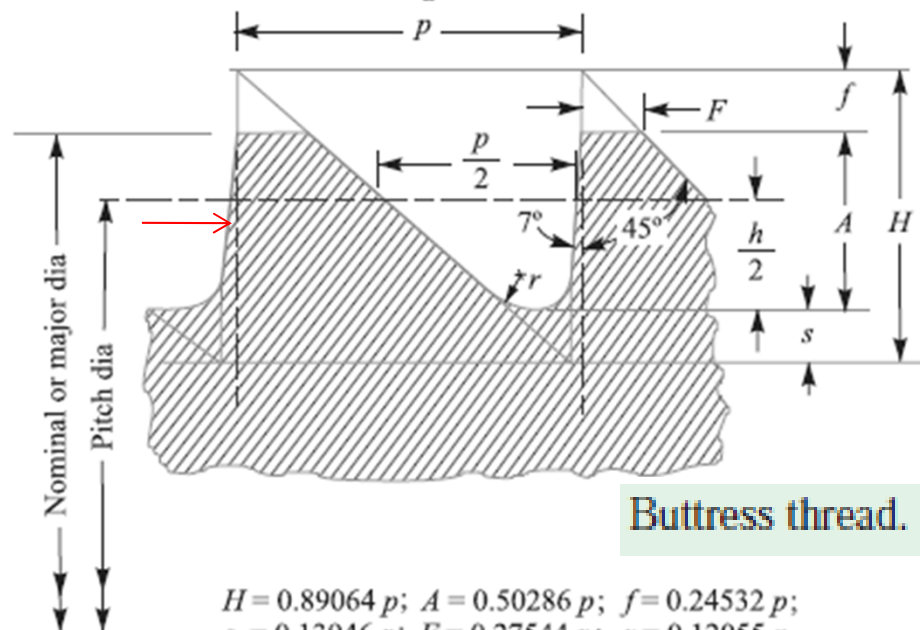
Square thread.



Acme thread.

The disadvantages of the Acme thread form are the much lower efficiency and the greater radial load on the nut, due to the thread angle

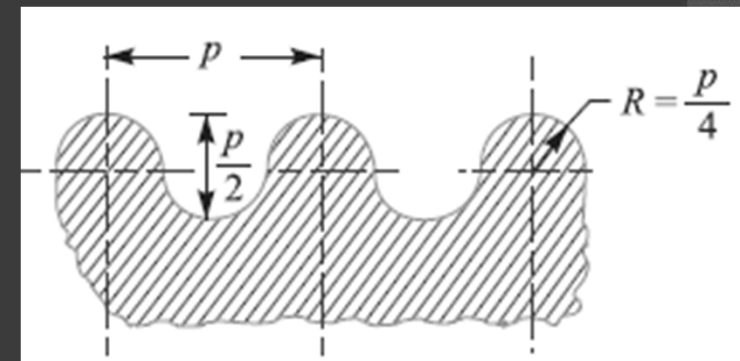
one direction only.



Buttress thread.

$$H = 0.89064 p; A = 0.50286 p; f = 0.24532 p;$$

$$s = 0.13946 p; F = 0.27544 p; r = 0.12055 p.$$



Knuckle thread.

What are the benefits of fine threaded fasteners over coarse threaded fasteners?

The potential benefits of fine threads are:

1. Fine thread is stronger than a coarse thread . This is both in tension (because of the larger stress area) and shear (because of their larger minor diameter).
2. Fine threads have also less tendency to loosen since the thread incline is smaller and hence so is the off torque.
3. Because of the smaller pitch they allow finer adjustments in applications that need such a feature.
4. Fine threads can be more easily tapped into hard materials and thin walled tubes.
5. Fine threads require less torque to develop equivalent bolt preloads.

On the negative side:

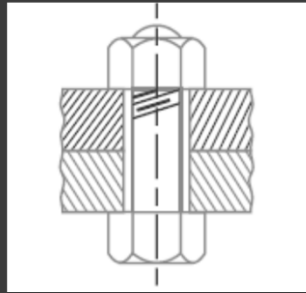
1. Fine threads are more susceptible to galling than coarse threads.
2. They need longer thread engagements and are more prone to damage and thread fouling.
3. They are also less suitable for high speed assembly since they are more likely to seize when being tightened.

Normally a coarse thread is specified unless there is an over-riding reason to specify a fine thread, certainly for metric fasteners, fine threads are more difficult to obtain.

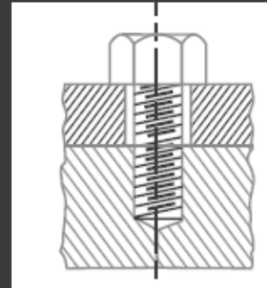
- We have a problem when tightening stainless steel bolts.
- They tend to seize.
- Whats happening?

Stainless steel can unpredictably sustain *galling (cold welding)*. Stainless steel self-generates an oxide surface film for corrosion protection. During fastener tightening, as pressure builds between the contacting and sliding, thread surfaces, protective oxides are broken, possibly wiped off, and interface metal high points shear or lock together. This cumulative clogging-shearing-locking action causes increasing adhesion. In the extreme, galling leads to seizing - the actual freezing together of the threads. If tightening is continued, the fastener can be twisted off or its threads ripped out.

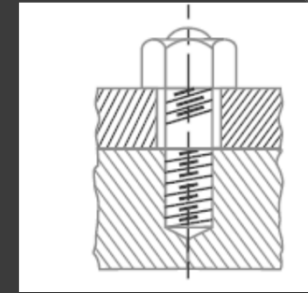
Common types of Screw Fastenings:



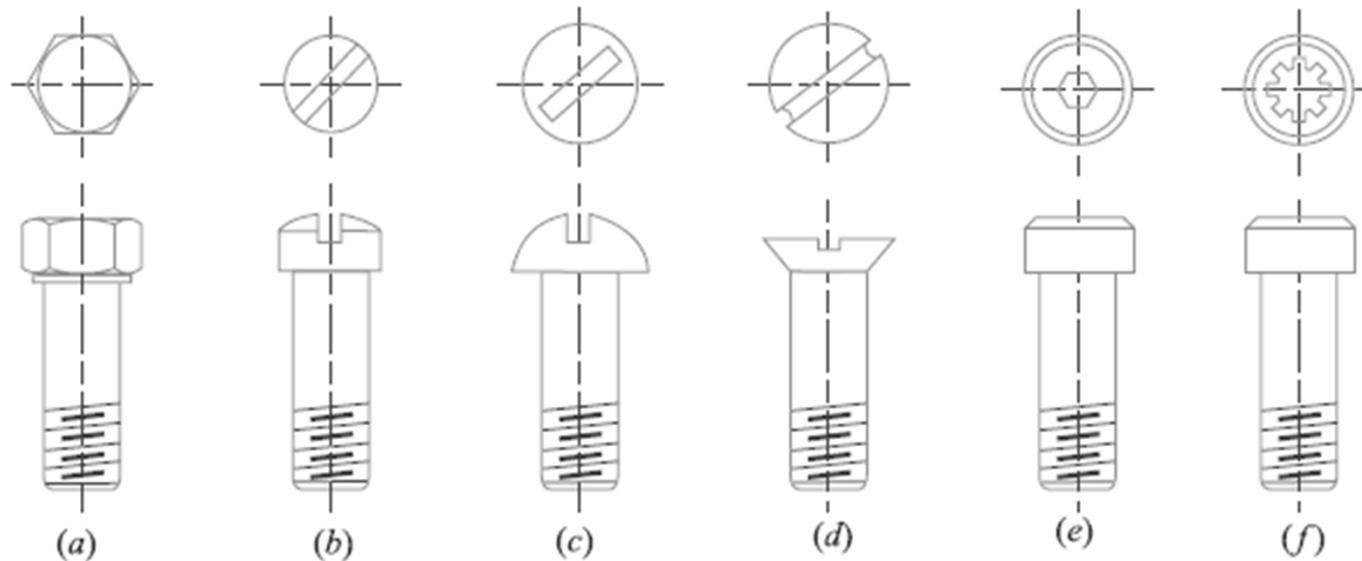
(a) Through bolt.



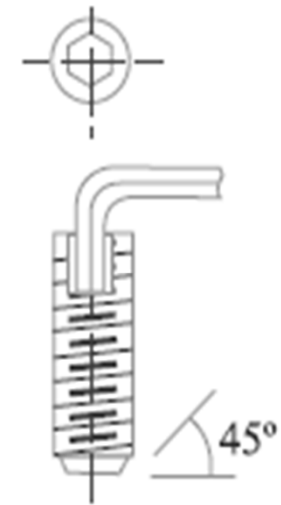
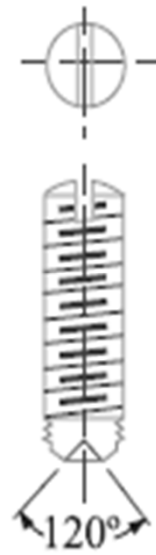
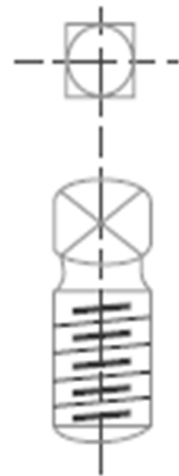
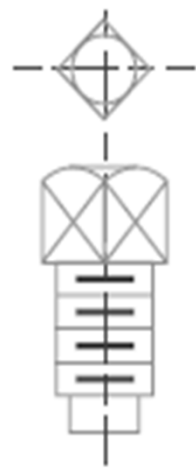
(b) Tap bolt.



(c) Stud.



(a) Hexagonal head; (b) Fillister head; (c) Round head; (d) Flat head;
(e) Hexagonal socket; (f) Fluted socket.



Set screws.

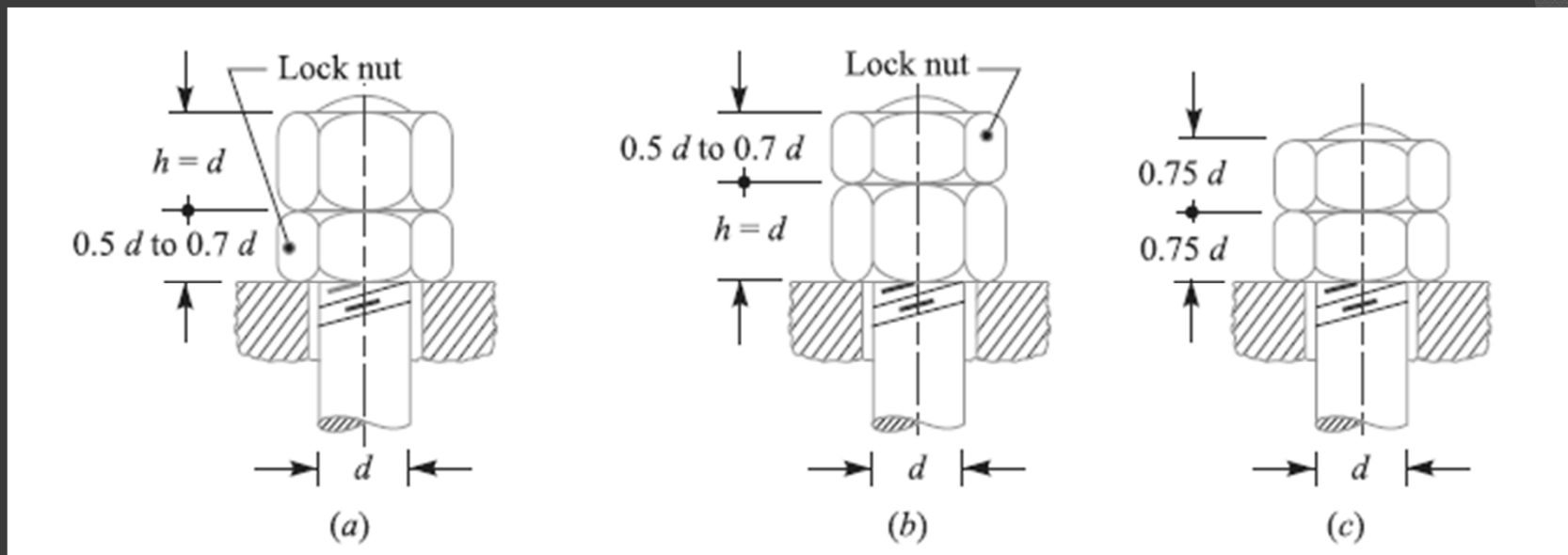
Locking Devices:

Ordinary thread fastenings remain tight under static loads, but many of these fastenings become loose under the action of variable loads.

This is dangerous

Q: What is the solution?

Ans: Locking Devices



Jam nut or lock nut.

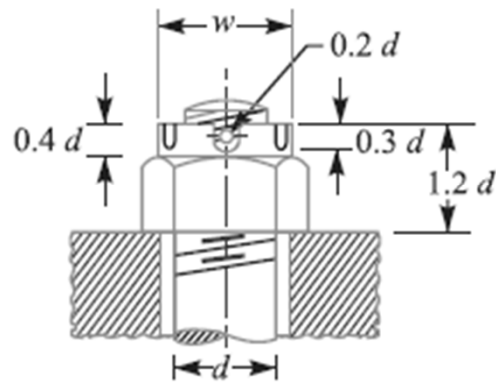


Fig. 11.16. Castle nut.

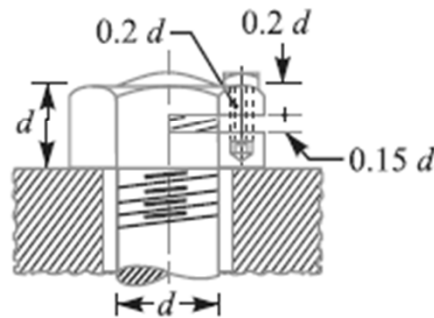


Fig. 11.17. Sawn nut.

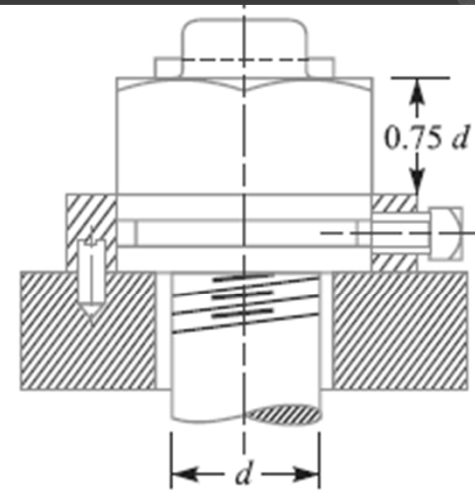


Fig. 11.18. Penn, ring or grooved nut.

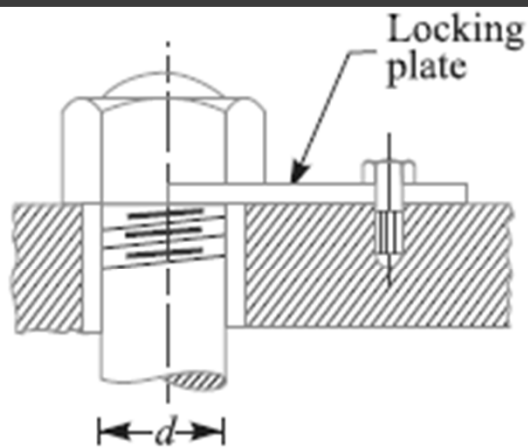


Fig. 11.20. Locking with plate.

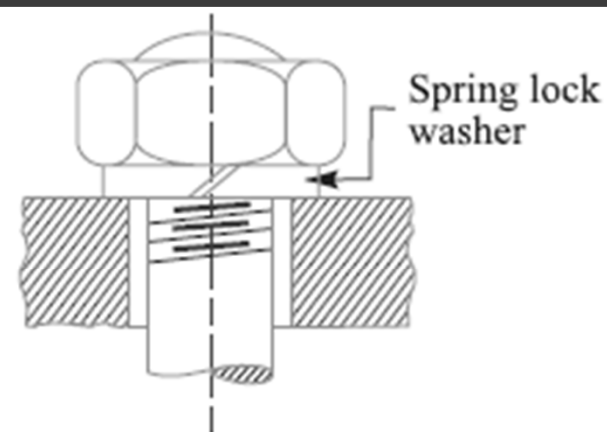
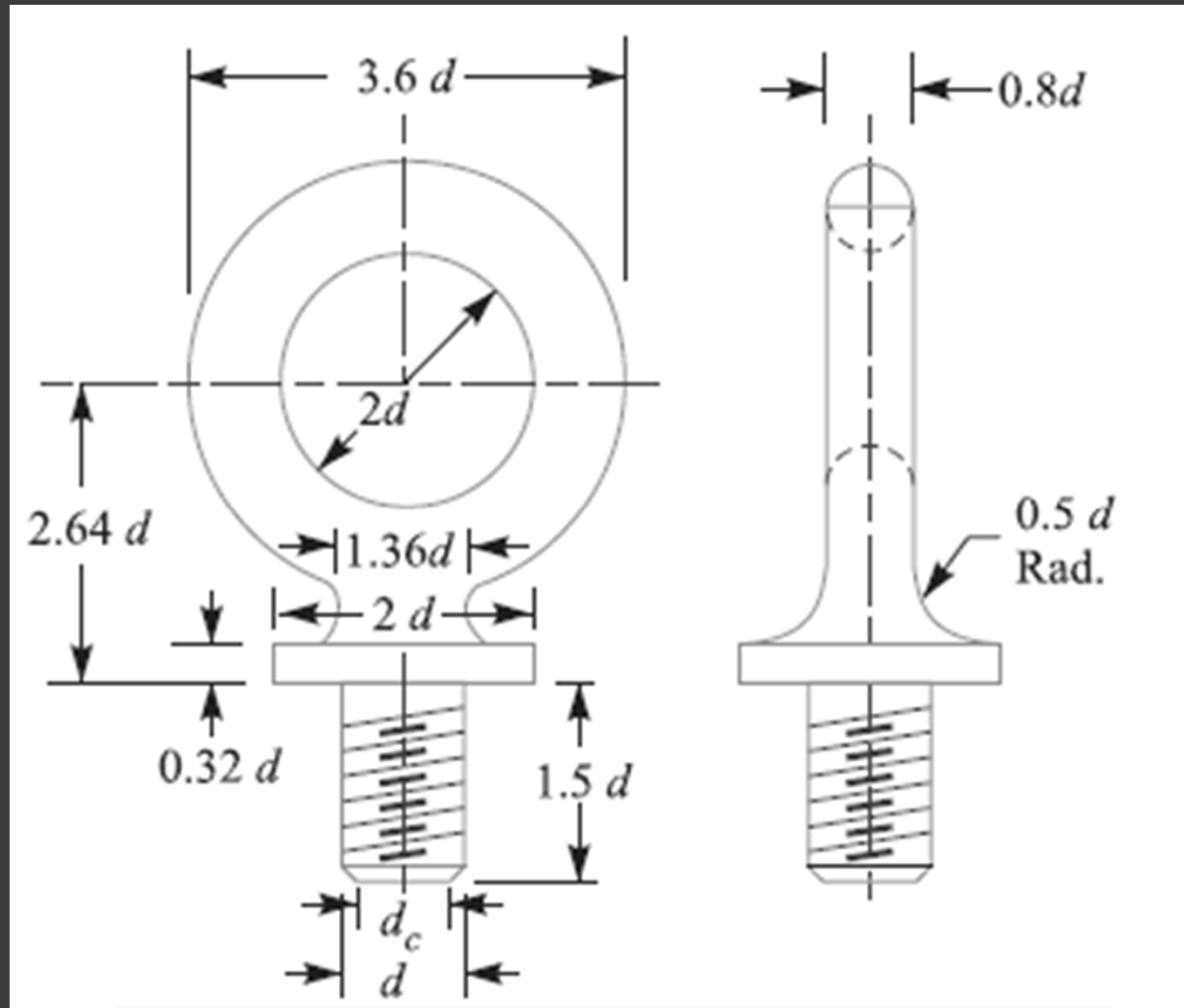
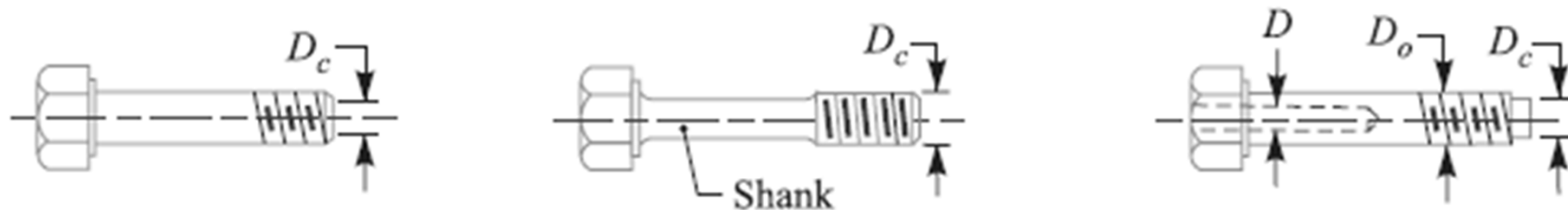


Fig. 11.21. Locking with washer.

Design of an Eye Bolt:



Bolts of Uniform Strength:



D = Diameter of the hole.

D_o = Outer diameter of the thread, and

D_c = Root or core diameter of the thread.

$$\frac{\pi}{4} D^2 = \frac{\pi}{4} [(D_o)^2 - (D_c)^2]$$

$$D^2 = (D_o)^2 - (D_c)^2$$

$$D = \sqrt{(D_o)^2 - (D_c)^2}$$

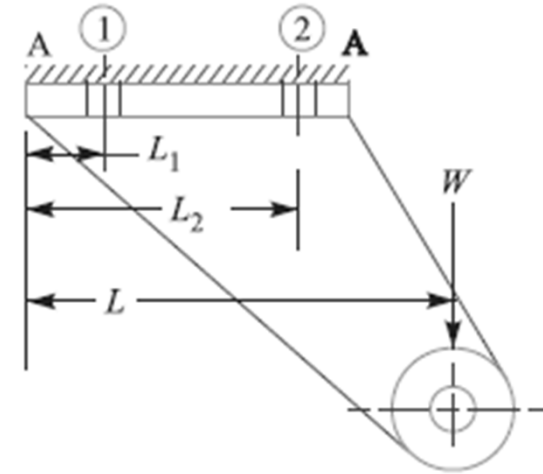
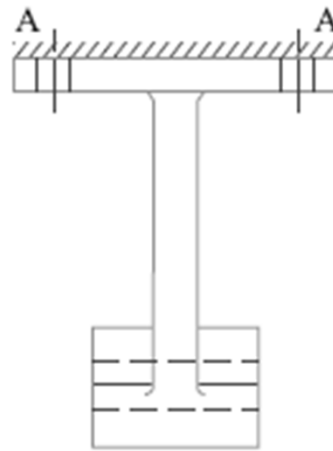
Design of Nut:

1. When the bolt and the nut is made of mild steel, the effective height of the nut is made equal to the nominal diameter of the bolt.
 2. If the nut is made of a weaker material than the bolt, then the height of nut should be larger, such as
 - 1.5 d - for gun metal
 - 2 d - C.I.
 - 2.5 d - aluminium alloys
- where d - nominal diameter of the bolt

Bolted Joints under Eccentric Loading:

1. Parallel to the axis of the bolts.
2. Perpendicular to the axis of the bolts.
3. In the plane containing the bolts.

Eccentric Load Acting Parallel to the Axis of Bolts:



direct tensile load

$$W_{t1} = \frac{W}{n}, \text{ where } n \text{ is the number of bolts.}$$

Let w be the load in a bolt per unit distance due to the turning effect of the bracket and let W_1 and W_2 be the loads on each of the bolts at distances L_1 and L_2 from the tilting edge.

$\therefore$ Load on each bolt at distance L_1 ,

$$W_1 = w.L_1$$

and moment of this load about the tilting edge

$$= w_1.L_1 \times L_1 = w (L_1)^2$$

Similarly, load on each bolt at distance L_2 ,

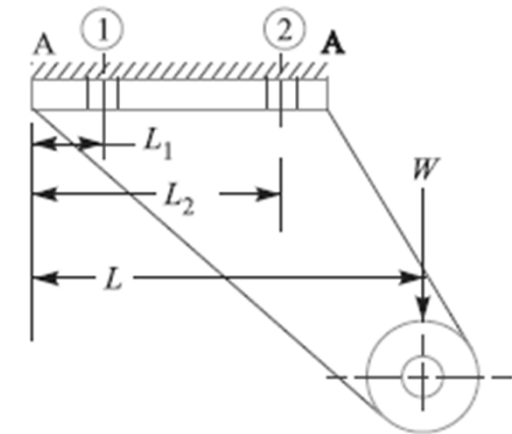
$$W_2 = wL_2$$

and moment of this load about the tilting edge

$$= wL_2 \times L_2 = w(L_2)^2$$

$\therefore$ Total moment of the load on the bolts about the tilting edge

$$= 2w(L_1)^2 + 2w(L_2)^2$$



Also the moment due to load W about the tilting edge

$$= WL$$

$$WL = 2w(L_1)^2 + 2w(L_2)^2 \quad \text{or} \quad w = \frac{WL}{2[(L_1)^2 + (L_2)^2]}$$

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance L_2 are heavily loaded.

$\therefore$ Tensile load on each bolt at distance L_2 ,

$$W_{t2} = W_2 = wL_2 = \frac{W.L.L_2}{2[(L_1)^2 + (L_2)^2]}$$

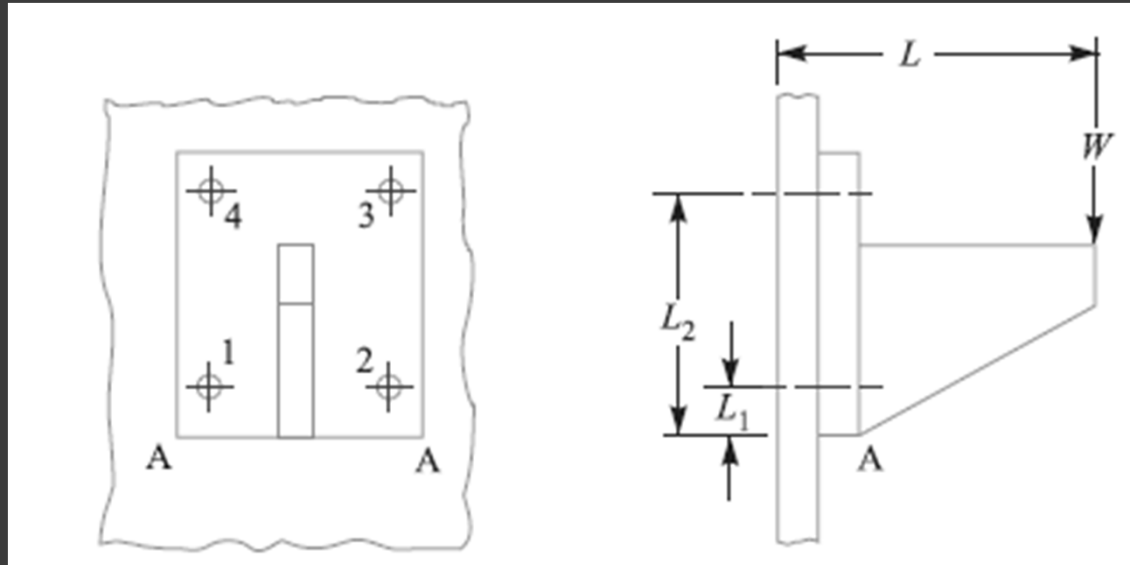
the total tensile load on the most heavily loaded bolt,

$$W_t = W_{t1} + W_{t2}$$

If d_c is the core diameter of the bolt and σ_t is the tensile stress for the bolt material, then total tensile load,

$$W_t = \frac{\pi}{4} (d_c)^2 \sigma_t$$

Eccentric Load Acting Perpendicular to the Axis of Bolts:



In this case, the bolts are subjected to direct shearing load which is equally shared by all the bolts. Therefore direct shear load on each bolts,

$$W_s = W/n, \text{ where } n \text{ is number of bolts.}$$

Maximum tensile load on bolt 3 or 4,

$$W_{\theta} = W_t = \frac{W.L.L_2}{2 [(L_1)^2 + (L_2)^2]}$$

Equivalent tensile load,

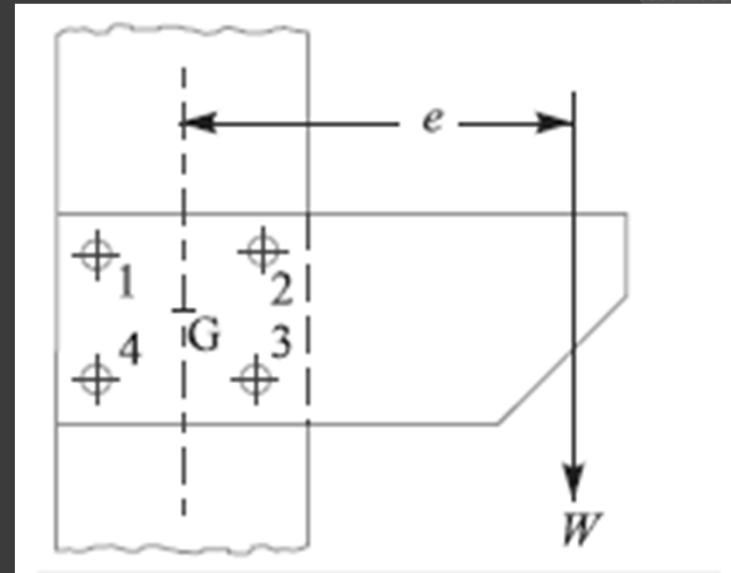
$$W_{te} = \frac{1}{2} \left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2} \right]$$

and equivalent shear load,

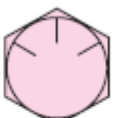
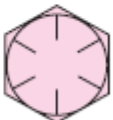
$$W_{se} = \frac{1}{2} \left[\sqrt{(W_t)^2 + 4(W_s)^2} \right]$$

Eccentric Load Acting in the plane Containing the Bolts:

Same procedure to be followed as discussed for eccentric loaded riveted joint.



SAE Specifications for Steel Bolts

SAE Grade No.	Size Range Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
1	$\frac{1}{4}$ – $1\frac{1}{2}$	33	60	36	Low or medium carbon	
2	$\frac{1}{4}$ – $\frac{3}{4}$	55	74	57	Low or medium carbon	
	$\frac{7}{8}$ – $1\frac{1}{2}$	33	60	36		
4	$\frac{1}{4}$ – $1\frac{1}{2}$	65	115	100	Medium carbon, cold-drawn	
5	$\frac{1}{4}$ –1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
5.2	$\frac{1}{4}$ –1	85	120	92	Low-carbon martensite, Q&T	
7	$\frac{1}{4}$ – $1\frac{1}{2}$	105	133	115	Medium-carbon alloy, Q&T	
8	$\frac{1}{4}$ – $1\frac{1}{2}$	120	150	130	Medium-carbon alloy, Q&T	
8.2	$\frac{1}{4}$ –1	120	150	130	Low-carbon martensite, Q&T	

ASTM Specifications for Steel Bolts

ASTM Designation No.	Size Range, Inclusive, in	Minimum Proof Strength,* kpsi	Minimum Tensile Strength,* kpsi	Minimum Yield Strength,* kpsi	Material	Head Marking
A307	$\frac{1}{4}$ – $1\frac{1}{2}$	33	60	36	Low carbon	
A325, type 1	$\frac{1}{2}$ –1	85	120	92	Medium carbon, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
A325, type 2	$\frac{1}{2}$ –1	85	120	92	Low-carbon, martensite, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
A325, type 3	$\frac{1}{2}$ –1	85	120	92	Weathering steel, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
A354, grade BC	$\frac{1}{4}$ – $2\frac{1}{2}$	105	125	109	Alloy steel, Q&T	
	$2\frac{3}{4}$ –4	95	115	99		
A354, grade BD	$\frac{1}{4}$ –4	120	150	130	Alloy steel, Q&T	
A449	$\frac{1}{4}$ –1	85	120	92	Medium-carbon, Q&T	
	$1\frac{1}{8}$ – $1\frac{1}{2}$	74	105	81		
	$1\frac{3}{4}$ –3	55	90	58		
A490, type 1	$\frac{1}{2}$ – $1\frac{1}{2}$	120	150	130	Alloy steel, Q&T	
A490, type 3	$\frac{1}{2}$ – $1\frac{1}{2}$	120	150	130	Weathering steel, Q&T	

Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs*

Property Class	Size Range, Inclusive	Minimum Proof Strength, [†] MPa	Minimum Tensile Strength, [†] MPa	Minimum Yield Strength, [†] MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	
4.8	M1.6–M16	310	420	340	Low or medium carbon	
5.8	M5–M24	380	520	420	Low or medium carbon	
8.8	M16–M36	600	830	660	Medium carbon, Q&T	
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	

What is bolt preload and why is it important?

Preload is the tension created in a fastener when it is tightened. This tensile force in the bolt creates a compressive force in the bolted joint known as clamp force. For practical purposes, the clamp force in an unloaded bolted joint is assumed to be equal and opposite of the preload. If proper preload, and thus clamp force, is not developed or maintained, the likelihood of a variety of problems such as fatigue failure, joint separation, and self-loosening from vibration can plague the bolted joint leading to joint failure.

